Nigerian Journal of Technology (NIJOTECH)
Vol. 36, No. 1, January 2017, pp. **110 – 118**

Print ISSN: 0331-8443, Electronic ISSN: 2467-8821
www.nijotech.com
http://dx.doi.org/10.4314/njt.v36i1.14

OPTIMAL EVAPORATING AND CONDENSING TEMPERATURES OF ORGANIC RANKINE CYCLE IN A HOT AND HUMID ENVIRONMENT

U. V. Ihuoma[1] and E. O. Diemuodeke[2,*]
[1,2] DEPT. OF MECHANICAL ENGR., COLLEGE OF ENGR., UNIVERSITY OF PORT HARCOURT, PORT HARCOURT, NIGERIA.
E-mail addresses: [1] *ihuomavicthor@gmail.com,* [2] *ogheneruona.diemuodeke@uniport.edu.ng*

ABSTRACT

This paper presents a thermodynamic analysis of an organic Rankine cycle (ORC) in a hot and humid environment. A theoretical procedure is proposed for the determination of the optimal evaporation temperature (OET) and optimal condensing temperature (OCT) of a subcritical ORC plant, which is based on thermodynamic theory; the heat input is selected as the objective function. The OETs and OCTs of 21 working fluids, which comprise wet, isentropic and dry fluids, are determined under given environmental conditions– hot and humid environment. The Engineering Equation Solver (EES) integrated development environment is used to optimize the heat input of the ORC plant. Results suggest that the wet fluids (namely R717, Methanol and Ethanol) have the potential of producing better thermodynamic figure of merit over the other fluid types.

Keywords: *Organic Rankine Cycle, Optimal Condensing Temperature, Optimal Evaporation Temperature, Hot and Humid Environment.*

1. INTRODUCTION

The desire to have access to energy has created negative multiplier effects, namely climate change, non-equilibrium of the ecosystem, pollution of the environment, and anxiety among nations. The use of organic fluids as the working fluid in the energy-to-power conversion system favour cycle efficiency [1]. Investigation has shown that low-grade waste heat accounts for more than half of the total heat generated in the industries [2]. The recovery of the waste-heat has generally been overlooked due to inefficient waste-heat-recovery methods, which is attributed to second law of thermodynamics. Besides the low-grade waste-heat from the industries, there are abundant low-grade energy sources that exist naturally, namely solar energy, biomass energy, and geothermal resources.

It has been reported that the global demand for primary energy is on the steady increase [3]; and that if the demand is maintained at a conservative average rate of 2%, the total global energy demand will increase by 100% in 30 years. The accelerated consumption of fossil fuels, if not abated would lead to a major health and energy crisis. However, the utilisation of low-grade energy has attracted appreciable attention due to its potential in relaxing environmental pollution and fossil fuel consumption [4]. In the recent years, there have been frantic scientific and political efforts to extend the market share of renewable energies and the utilisation of low-grade heat in the energy mix [5]–[11].

Organic Rankine Cycles (ORCs) have long been investigated for the production of a non-negligible power [12].Conceptually, the Organic Rankine Cycle (ORC) is similar to the steam Rankine Cycle (SRC), since both are based on the vaporisation of high pressure liquid working fluid in a conduit; and, thereafter, the working fluid in vapour form is expanded in a work-producing expansion device to produce a shaft work (mechanical work) [13]. The ORC plant has a simpler system configuration, as compared to a SRC, in that: absence of drum-boiler connection and presence of a single heat exchanger for the three evaporation phases: preheating, vaporisation and superheating [14]. Figure 1shows a solar thermal driven ORC. Besides the main configuration, a recuperator is desired to be included in the system configuration to preheat liquid between the pump outlet and the expander outlet. The commercially available ORC plants are mainly based on the simple configuration, which is normally adapted for optimisation [14], [15]. It should be noted that advances have been made on the simple ORC configuration, which are currently being studied for proof-of-concepts.

*** Corresponding author, tel: +234 – 805 – 632 – 0209**

The heat input to the ORCplants is dependent on a wide range of heat sources namely waste heat from the condenser of a conventional or a nuclear power plant, waste heat from industrial processes, solar radiation, and geothermal energy. The solar power plant has a well-proven technology[16]: the solar radiation is reflected onto a collector, transferring heat from the collector to an intermediate fluid at high temperature, and transferring heat from the intermediate fluid to the power plant working fluid. The solar powered ORC power plants are a promising technology, since it has been reported to have a decrease investment costs at small scale– it features lower evaporation temperature and a kW levels scaled down installation power capacity [14]. However, the size of the solar collector is proportional to the heat requirement of the ORC plant. This implies that minimisation of the heat input and maximising the ORC plant's figure of merit is highly essential.

The working fluid is an important factor to be considered in the design and optimization of an organic Rankine cycle for various sources of low-grade heat. Zhaolin and Haruki [17]discussed the performance of supercritical cycles for geothermal binary design. Propane, R-125, and R-134a were considered as working fluids for a given geothermal design and their thermodynamic parameters were calculated. Propane and R-134a emerged as the most suitable working fluid of supercritical cycles of the particular geothermal source because of their higher power output when compared to R-125.

On the other hand, Saleh et al.[18]performed a thermodynamic analysis of 31 pure working fluids using the BACKONE equation of state for ORC geothermal plant. The working fluids operated between 100 and 30 °C evaporator and condenser temperatures, respectively. The study showed that the highest thermal efficiencies are obtained with dry fluids in subcritical cycles with regenerator. The pinch analysis was also performed for the heat transfer between the source and the working fluid and reported that the largest amount of heat can be transferred to a super-critical fluid and the least to a high-boiling subcritical fluid. Xu and He [19]performed a comparison between a vapour injector-based novel regenerative ORC and the simple ORC configuration; the analysis was based on thermal efficiency and power output. The results indicate that the regenerative ORC has a better performance over the simple ORC configuration. However, the study further showed that the evaporating temperature (ET) and condensing temperature (CT) affect the thermal performance and net power output of the cycles. Dai et al.[20]conducted parametric optimisation of ORC with exergy efficiency.

He *et al.* [21] considered the optimisation of a simple configuration ORC plant for 28 working fluids; the evaporating temperature was selected as the decision variable and the net power output served as the objective function. The study devised an analytical procedure for the determination of optimum evaporating temperature, which maximizes the net power output, of a subcritical cycle without superheating. Results of the study were in agreement with a simplified formula for determining the OET. It was established that the net power output was larger for working fluids whose critical temperature is close to that of the heat source. For heat source and sink temperatures equal to 150 °C and 20 °C, respectively, the authors concluded that R114, R245fa, R123, R601a, n-pentane, R141b and R113 are the most suitable working fluids. However, the condensing temperature was fixed in the analysis, which undermines the influence of ambient conditions on the ORC performance.

The generation of power using low-grade heat sources has seen increased investigations in recent years and has seen considerable implementation and improvements. However, it is observed that exhaustive investigation into the effect of the optimum evaporating and condensing temperatures of the working fluid on the Organic Rankine Cycle has not been completely investigated; even the ambient conditions have not been integrated in many of the optimised systems. This work focuses on the determination of optimum evaporating and condensing temperatures of selected working fluids by considering the ambient conditions.

2. MATERIALS AND METHODS

2.1System Description

A simple configuration of a solar thermal driven ORC plant, as investigated in this paper, is shown in Figure 1. This configuration consists of a working fluid pump, an evaporator driven by heat from a solar collector, an expander, and an air cooled condenser. Working fluid with low boiling point is pumped to the evaporator, where it is heated and vaporized by heat from a solar collector. The solar collector heats water which in turn transfers heat to the working fluid in the evaporator. The generated high pressure vapours flows into the expander, and is converted into shaft work. Simultaneously, the shaft work from the expander drives the generator and electric energy is generated. Thereafter, the vapour exits the expander as low pressure vapour and is led to the condenser where it is condensed by the cooling air. The condensed working fluid is pumped back to the evaporator and a new cycle begins.

2.2 Thermodynamic Analysis of ORC

Fig. 2(a) illustrates the ideal thermodynamic processes on the T-s diagram for this ORC system. Generally, there are four different processes: Expansion process 1-2; Isobaric condensation process 2-3; Pumping process 3-4 and Isobaric heat absorption process 4-1.

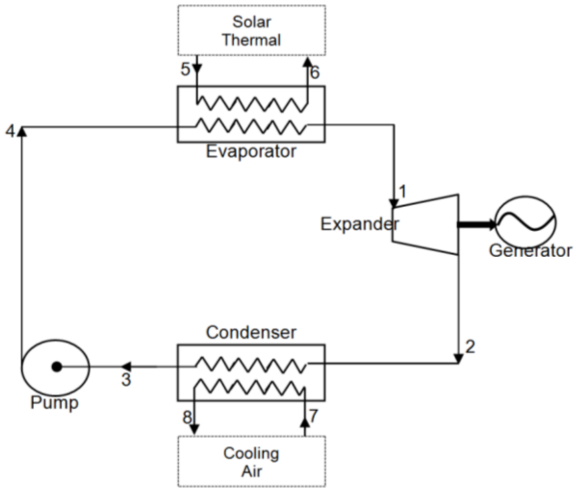

Figure 1: Schematic Diagram of Organic Rankine Cycle

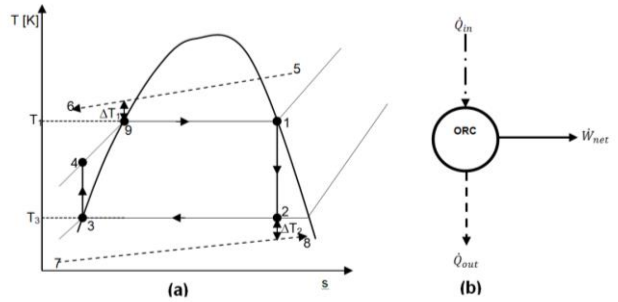

Figure 2: Thermodynamic Processes: (a) T-sdiagram (b) Energy balance

The ORC specifications considered in this paper, for the heat source (solar-thermal collector) and the pertinent environmental conditions, are given in section three. The pertinent parameters are specified to utilise the available solar irradiance through the solar-thermal collector for maximum net power output in the geographical location considered. The net power output of the ORC reflects its capability to make use of the heat obtained from the solar collector. The heat input is selected as the objective function in this paper. The evaporation and condensing temperatures that give the maximum net power output are defined as the optimal evaporating temperature (OET) and optimal condensing temperature (OCT), respectively.

The theoretical formula of the OET and OCT using the heat input as the objective function is derived based on thermodynamic theory.

Important assumptions are made, according to reference [21], to simplify the thermodynamic analysis of the system. The assumptions are:

i. Heat losses and mechanical losses are negligible.
ii. The units of the system are treated as control volumes.
iii. No pressure drop in the evaporator, pipes and condenser.
iv. The components are on the same datum level at inlet and exit of the fluid and the variation in the velocity and altitude are negligible. Hence, the changes in kinetic and potential energies are neglected.
v. Mass flow and energy flow through the plant were assumed to be in steady state.
vi. Isentropic efficiencies of the pump and the expander are given as 0.75 and 0.80, respectively.
vii. The working fluid at the expander inlet and condenser outlet is saturated vapour and saturated liquid, respectively.

The net power output of the ORC is expressed as:

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_p \quad (1)$$

where $\dot{W}_t$ and $\dot{W}_p$ are the power generated by the expander and power consumed by the pump, respectively.

The power generated by the expander is expressed as:

$$\dot{W}_t = \dot{m}_{wf}\,(h_1 - h_2)\eta_s\eta_G \quad (2)$$

where h_1 and h_2 are the specific enthalpies of the working fluid at the inlet and outlet (for an ideal process) of the expander, respectively;η_s and η_G are the expander isentropic efficiency and the generator efficiency, respectively, and $\dot{m}_{wf}$is the mass flow rate of the working fluid.

The power consumed by the pump can be expressed as:

$$\dot{W}_p = \frac{\dot{m}_{wf}(h_4 - h_3)}{\eta_p} \quad (3)$$

where $\eta_p$is the isentropic efficiency of the pump;h_4 is the ideal specific enthalpy of the working fluid at the outlet of the pump; and h_3 is the specific enthalpy of the working fluid at the outlet of the condenser.

Figure 2(b) shows the heat and work interactions of the ORC plant, which can be expressed mathematically as:

$$\dot{Q}_{in} = \dot{W}_{net} + \dot{Q}_{out} \quad (4)$$

where $\dot{Q}_{in}$and $\dot{Q}_{out}$are the heat input and heat rejected, respectively; and$\dot{W}_{net}$represents the net power output.

The heat rejected is expressed as:

$$\dot{Q}_{out} = \dot{m}_a \times (c_{Pd.a} + g c_{Pw.v}) \times (T_3 - \Delta T_2 - T_7) \quad (5)$$

$\dot{m}_a$is the mass flow rate of air;$c_{pd.a}$is the specific heat capacity of dry air;$c_{pw.v}$is the specific heat capacity of water vapour;g is the humidity ratio;T_3,ΔT_2 and $T_7$are the condensing temperature, condenser pinch temperature and temperature of cooling air (the ambient temperature).

The net power output of the ORC can also be expressed as:

$$\dot{W}_{net} = \dot{m}_{wf}(w_t - w_p) \quad (6)$$

where w_t and w_p are the specific work produced by the expander and specific work input into the pump, respectively.

From Figure 2, the specific work output can be determined by the area of 1-2-3-4-9-1, $A_{1-2-3-4-9-1}$; whereas the specific work input into the pump can be represented as the 3-4-9-3, $A_{3-4-9-3}$. It can be shown that $A_{1-2-3-4-9-1} \gg A_{3-4-9-3}$[21], therefore, $w_t - w_p \sim = A_{1-2-3-4-9-1}$. Thus, the specific net power output of the ORC plant could be expressed as:

$$w_t - w_p = \left[(T_1 - T_3)(s_1 - s_9) + \frac{1}{2}(T_1 - T_3)(s_9 - s_3)\right]\eta_s\eta_G \quad (7)$$

where T_1 and T_3 are the evaporating temperature and condensing temperature, respectively; s_1 and s_3 are the specific entropies of the working fluid at the inlet of the expander and pump, respectively; and s_9 is the specific entropy of the saturated liquid working fluid at the temperature of T_1.

The mass flow rate of the working fluid is obtained, based on the preceding approximation, by:

$$\dot{m}_{wf} = \frac{c_{ph}\dot{m}_h(T_5 - T_1 - \Delta T_1)}{\mu} \quad (8)$$

c_{ph} is the specific heat capacity of the heat source at constant pressure; $\dot{m}_h$ is the mass flow rate of heated fluid in the solar-thermal collector; T_5 and ΔT_1 are the inlet temperature of the heat from the solar collector and evaporator pinch temperature, respectively; and μ is the latent heat of working fluid.

Thermodynamically, the entropy change for isobaric process can be expressed as $ds \equiv dQ/T$, where ds is entropy change, dQ is change in heat and T is temperature. Therefore, the latent heat can be expressed accordingly as[21]:

$$\mu = T_1(s_1 - s_9) \quad (9)$$

The entropy change of the working fluid from state point 3 to state point 9 can be expressed as[21]:

$$s_9 - s_3 \approx c_{pl} ln\frac{T_1}{T_3} \quad (10)$$

c_{pl} is the specific heat capacity of the working fluid at constant pressure.

Substituting Eqns (7) through (10) into (6)gives an approximate expression for the net power output of the ORC, thus,

$$\dot{W}_{net} \approx \frac{c_{ph}\dot{m}_h\eta_s\eta_G(T_5 - T_1 - \Delta T_1)}{T_1}(T_1 - T_3)\left(1 + \frac{c_{pl}T_1}{2\mu}\ln\frac{T_1}{T_3}\right) \quad (11)$$

Substituting Eqns. (5) and (11) into Eqn. (4) gives an approximate expression for the heat input into the ORC, which is expressed as:

$$\dot{Q}_{in} = (\dot{m}_a \times (C_{Pd.a} + gC_{Pw.v}) \times (T_3 - \Delta T_2 - T_7)) + \frac{C_{ph}\dot{m}_h\eta_s\eta_G(T_5 - T_1 - \Delta T_1)}{T_1} \times (T_1 - T_3)\left(1 + \frac{C_{pl}T_1}{2\mu}\ln\frac{T_1}{T_3}\right) \quad (12)$$

The empirical relation for latent heat of working fluid proposed by Kobayashi [22], as presented by reference[21],sufficed:

$$\mu = R_gT_c[7.08\left(1 - \frac{T_1}{T_c}\right)^{0.354} + 10.95\omega\left(1 - \frac{T_1}{T_c}\right)^{0.456} \quad (13)$$

R_g is the specific gas constant and T_c is the critical temperature of the working fluid. ω(-) is the acentric factor of the working fluid.

Differentiating the Eqn.(12) with respect to T_1 and T_3 and equating to zero gives the theoretical framework for obtaining the OET and OCT as

$$\frac{d\dot{Q}_{in}}{dT_1} = \frac{T_3(T_5 - \Delta T_1)}{{T_1}^2} - 1 + \frac{c_{pl}}{2\mu}\left((T_5 - \Delta T_1 + T_3)\left(1 + \ln\frac{T_1}{T_3}\right) - T_1\left(2\ln\frac{T_1}{T_3} + 1\right) - \frac{T_3(T_5 - \Delta T_1)}{T_1} - \frac{1}{\mu}\frac{\partial\gamma}{\partial T_1}(T_5 - T_1 - \Delta T_1)(T_1 - T_3)\ln\frac{T_1}{T_3}\right) = 0 \quad (14)$$

$$\frac{d\dot{Q}_{in}}{dT_3} = \dot{m}_a(c_{pd.a} + gc_{pw.v}) - C_{ph}\dot{m}_h\eta_s\eta_G\frac{(T_5 - T_1 - \Delta T_1)}{T_1} \times \left(1 + \left(\frac{C_{pl}T_1}{2\mu}\ln\frac{T_1}{T_3} + \frac{C_{pl}T_1T_3}{2\mu}\frac{\partial}{\partial T_3}\left(\ln\frac{T_1}{T_3}\right)\right)\right) = 0 \quad (15)$$

Where

$$\frac{\partial\mu}{\partial T_1} = -R_g[2.51\left(1 - \frac{T_1}{T_C}\right)^{-0.646} + 4.99\omega(1 - \frac{T_1}{T_C})^{-0.544}]$$

The first law thermodynamic efficiency of the system for each working fluid is determined by dividing the net power output by the heat input, which could be expressed as:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \quad (16)$$

2.3 Choice of Working Fluids

Factors such as thermodynamic performance, stability, non-fouling, non-corrosiveness, non-toxicity, non-flammability of the working fluid are important in the design of an ORC. In this paper, the focus is on maximum net power, minimum heat input and efficiency as the criteria to screen the working fluids for the ORC.

The slope of the saturated vapour curve on T-s diagram (dT/ds) is used to classify the working fluids as wet, isentropic or dry fluids. The parameter ξ, which can be defined as the inverse of the slope of the saturated vapour curve on the T-s diagram, is used to classify the working fluids as follows: $\xi < 0$: a wet fluid; $\xi \sim 0$: an isentropic fluid; and $\xi > 0$: a dry fluid. Accordingly, the Table 1 presents the various properties of the working fluids considered for the ORC plant[21]. A total of 21 working fluids are considered in the analysis of the OET and OCT of the ORC plant.

Table 1: Properties of the working fluids[21]

S/No	Working Fluids	Fluid Type	Mol. Weight [g/mol]	Critical Temp [K]
1	R600a	Dry	58.12	407.8
2	R601a	Dry	72.15	460.4
3	n-Pentane	Dry	72.15	469.7
4	n-Hexane	Dry	86.17	507.9
5	n-Heptane	Dry	100.2	540.1
6	n-Octane	Dry	114.2	569.3
7	n-Dodecane	Dry	170.3	658.1
8	n-Decane	Dry	142.28	617.7
9	R142b	Isentropic	100.5	410.3
10	R114	Isentropic	170.92	418.9
11	R600	Isentropic	58.12	425.1
12	R245fa	Isentropic	134.05	427.2
13	R123	Isentropic	152.93	456.8
14	R11	Isentropic	137.37	471.2
15	R141b	Isentropic	116.95	477.4
16	R113	Isentropic	187.38	487.3
17	Toluene	Isentropic	92.14	591.8
18	Cyclohexane	Isentropic	84.16	553.6
19	R717	Wet	17.03	405.4
20	Methanol	Wet	32.04	513.4
21	Ethanol	Wet	46.07	513.9

3. RESULTS AND DISCUSSION

This section presents the results from the OET and OCT analysis based on minimum heat input requirement. Eqs (14) and (15) are solved simultaneously using the Engineering Equation Solver (EES) platform, which is able to handle system of nonlinear equations by quadratic approximation method. The thermodynamic properties of various working fluids are automatically evaluated by the EES.

The input parameters of the plant are presented in Table 2. Some of the data were obtained from EES fluid property library, some are constraints imposed on the system, the average ambient temperature and relative humidity data obtained for Port Harcourt climatic zone from the Nigerian Meteorological Agency (NIMET) [23]for a period of 11 years. The ambient data represents the hot and humid conditions, which are typical of tropical environments. Other pertinent specifications are presented in Table 3.

3.1 OET and OCT Results

The results of the simulation using EES and the input data, as shown in Table 2, are presented in Table 3. The optimal evaporating temperature and the optimal condensing temperature from the current analysis are presented in the Table 3 with their corresponding values from established previous works [21], [24]. The relative deviations of the results from the results of the previous works are quite high. This is not surprising, as previous works did not consider hot and humid conditions that are considered in this current analysis. Moreover, all the previous works fixed the condensing temperature, which may impact significantly on the optimal performance of the ORC plant. Therefore, direct comparison of results with established data was not possible as there are no available established works that simultaneously considered the evaporating temperature, condensing temperature and environmental conditions.

Table 2 Specifications of the ORC Plant Condition and other Pertinent Parameters.

S/No	Parameter	Symbol	Value	Unit
1	Heat source temperature	T_5	423.15	K
2	Mass flow rate of heat source fluid	$\dot{m}_h$	1	kg/s
3	Mass flow rate of cooling air	$\dot{m}_a$	5	kg/s
4	Average ambient temperature	T_7	301.15	K
5	Environment pressure	P_1	101.3	kPa
6	Pinch temperature difference in evaporator	ΔT_1	5	K
7	Pinch temperature difference in condenser	ΔT_2	5	K
8	Isentropic efficiency of the expander	η_s	0.80	-
9	Generator efficiency	η_G	0.96	-
10	Pump isentropic efficiency	η_p	0.75	-
11	Average relative humidity	R_h	0.82	-
12	Specific heat capacity of air	$c_{pd.a}$	1.007	kJ/kgK
13	Specific heat capacity of Water Vapour	$c_{pw.v}$	4.183	kJ/kgK
14	Cooling air mass flow rate/heat source fluid mass flow rate	$\dot{m}_a/\dot{m}_h$	5.00	-

Table 3: OET and OCT Results for Selected Working Fluids

S/No	Working Fluids	Fluid Type	OET [K]	OCT [K]	OET[a] [K]	OET[b] [K]	RD[a] %	RD[b] %
1	R600a	Dry	388.6	357.0	364.8	366.2	6.52	6.12
2	R601a	Dry	386.1	352.6	362.4	362.9	6.54	6.39
3	n-Pentane	Dry	390.0	360.0	362.0	362.9	7.73	7.47
4	n-Hexane	Dry	388.8	358.0	361.6	361.6	7.52	7.52
5	n-Heptane	Dry	387.7	356.0	361.4	360.9	7.28	7.43
6	n-Octane	Dry	387.3	355.3	361.3	360.6	7.20	7.40
7	n-Dodecane	Dry	387.2	355.2	361.2	360.4	7.20	7.44
8	n-Decane	Dry	387.1	355.0	361.2	360.3	7.17	7.44
9	R142b	Isentropic	389.6	357.9	363.3	366.7	7.24	6.24
10	R114	Isentropic	392.8	364.5	364.8	365.6	7.68	7.44
11	R600	Isentropic	388.6	357.0	363.2	365.3	6.99	6.38
12	R245fa	Isentropic	388.9	357.4	363.7	365.0	6.93	6.55
13	R123	Isentropic	382.8	346.6	361.7	362.6	5.83	5.57
14	R11	Isentropic	375.5	333.3	360.0	362.4	4.31	3.61
15	R141b	Isentropic	385.6	352.0	360.6	360.4	6.93	6.99
16	R113	Isentropic	388.6	357.6	361.5	361.6	7.50	7.47
17	Toluene	Isentropic	378.9	339.7	359.1	362.1	5.51	4.64
18	Cyclohexane	Isentropic	382.1	345.6	359.9	360.5	6.17	5.99
19	R717	Wet	364.3	312.4	360.3	363.7	1.11	0.16
20	Methanol	Wet	363.4	312.5	357.0	360.6	1.79	0.78
21	Ethanol	Wet	369.7	323.0	357.8	359.2	3.33	2.92

[a] OET obtained by a simplified model [24]at constant OCT, without the effect of relative humidity, and corresponding relative deviation (RD)

[b] OET obtained at constant OCT, without the effect of relative humidity, and corresponding relative deviation [21].

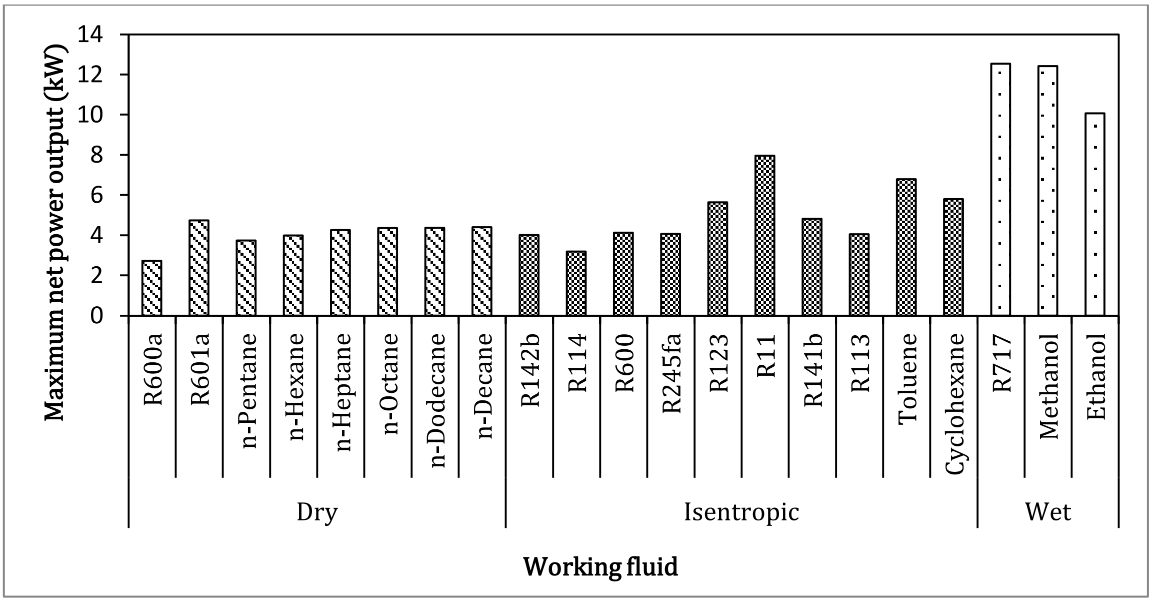

Figure 3: Maximum Net Power Output

Figures 3, 4and 5 show the maximum net power output, minimum heat input and system efficiency of the various working fluids. Higher net power output means that more power could be obtained under the same condition of heat source and ambient conditions.

The maximum net power output values vary with different working fluids as seen in Figure3. The figure suggests that the wet working fluids (namely R717, Methanol and Ethanol) present the best net power output, under the same operating conditions. Therefore, it is shown that the wet fluids have the potential of producing high power over the other fluid types considered (isentropic and dry). The observation may be as a result of the low condensing temperatures the wet fluids feature, under the ambient operating conditions. The highest net power of the ORC is about 12.54 [kW] for R717, and the lowest net power output of the ORC is 2.73 [kW] for R600a.

From Figure 4, the R600a requires the largest heat input; whereas the R717 requires the least heat input and lesser surface area of solar collector. More so, the fluids that produce the maximum net power output require the least heat input; such fluids tend to have greater system efficiency when used in the ORC, as shown in Figure5.

To justify the consideration of hot and humid environment conditions in this current work, system simulation was done under the hot and humid environment conditions, and the ISO standard conditions. According to ISO3997, the ISO standards temperature and relative humidity are 288.15 K and 60%, respectively. The heat input which is the objective function in this paper is considered for each working fluid at ISO conditions and compared with the heat input at hot and humid environment conditions, as shown in Figure 4.

It is observed from Figure 4 that additional heat input is required under the ISO conditions, in comparison with the hot and humid conditions, to produce the same amount of power. The excess heat input of the ISO conditions over the hot and humid conditions indicates that the ORC plant under the ISO operating conditions will be economically inferior to the hot humid and conditions considered, since the heat generated is dependent on the solar-thermal collector surface area. This supports the necessity to consider the hot and humid conditions, which have not been fully considered previously. It also lends further credibility to the thermodynamic formulas formulated in this paper to optimize the heat input to an ORC under the hot and humid environments.

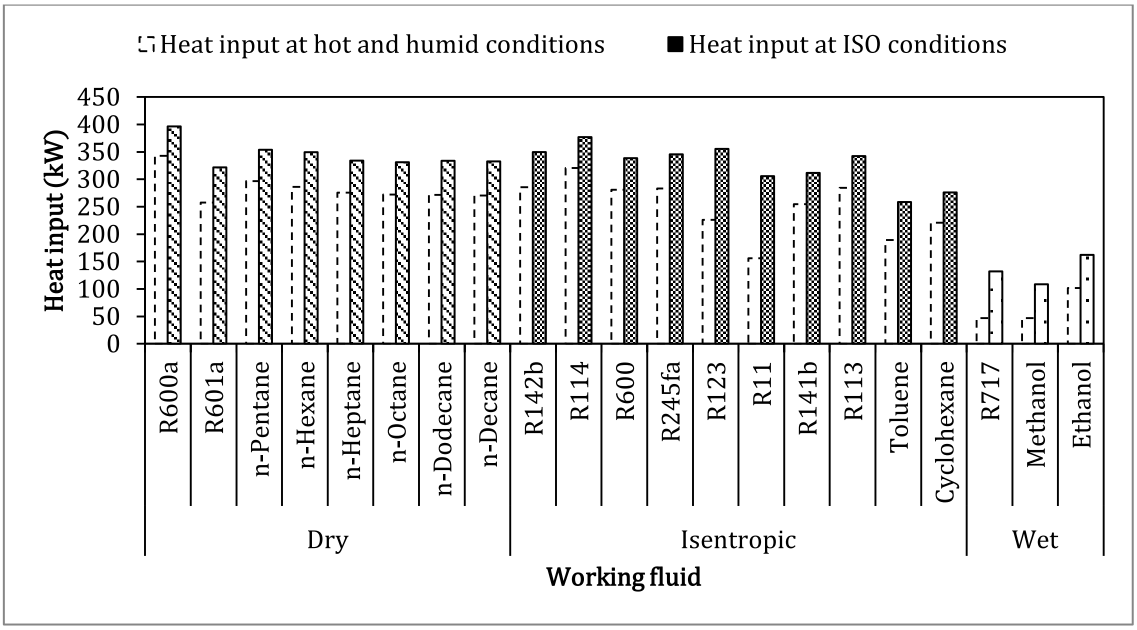

Figure 4 Heat Input for Selected Working Fluids

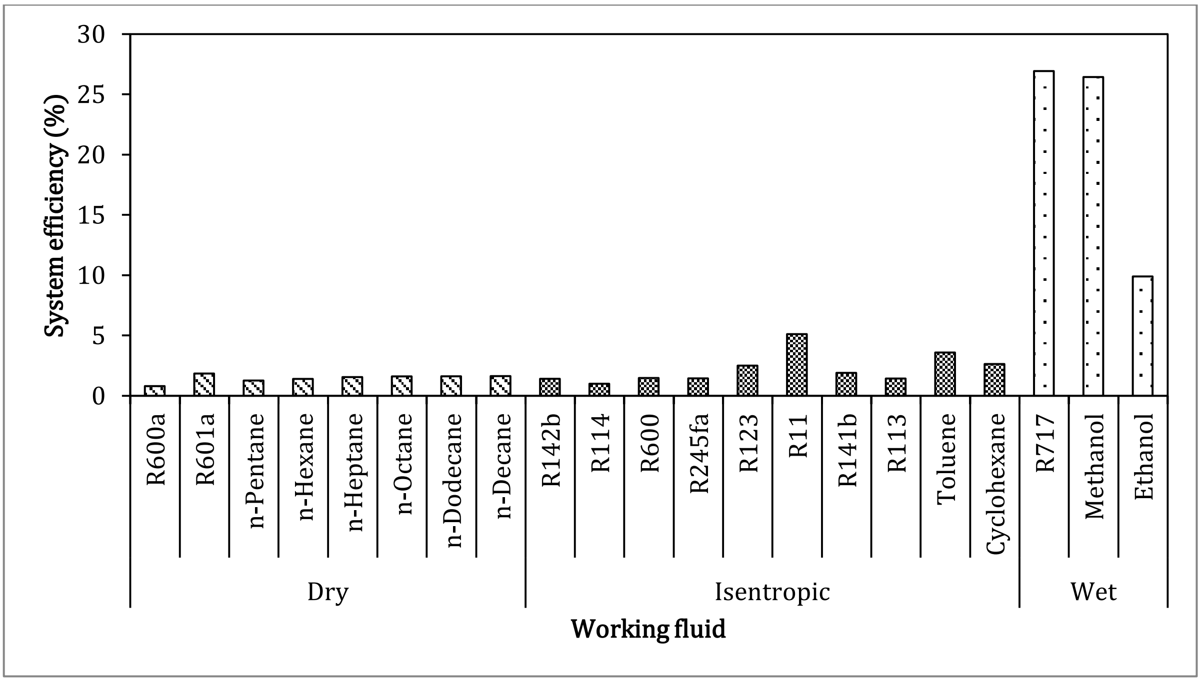

Figure 5: System Efficiency of ORC for Selected Working Fluids under Hot and Humid Environmental Conditions

Figure 5 indicates that R717 has the highest system thermal efficiency of 26.9% under the hot and humid environmental conditions, closely followed by Methanol; whereas R600a has the least system efficiency of about 0.8%, closely followed by R114. However, the theoretical net power is 12.54 kW at 26.9% thermal efficiency for R717 working fluid, which shows that the utilisation of low grade energy sources (e.g. solar thermal, ocean thermal and biomass waste) have the potential for electricity generation.

It is observed from the simulation that the ratio of the mass flow rate of the cooling air ($\dot{m}_a$) to the mass flow rate of the heat source fluid ($\dot{m}_h$) has a significant effect on the OET, OCT and net power output, as shown in the Figures 6 and 7. It means that, outside the ambient conditions, the mass flow rates of the working fluid and heat source fluid have control over the OET and OCT. It is, therefore, necessary to establish the mass flow ratio to determine the practical operating OET and OCT. Typical practical operating mass flow ratios range from 4.5 to 5.3 [25]; however, for the purpose of this paper the ratio 5.0 was adopted. R717 was selected to further illustrate the significance of this observation.

Figure 6 shows that the OET and OCT of R717 decrease as the ratio of the mass flow rate of air ($\dot{m}_a$) to mass flow rate of heat source fluid ($\dot{m}_h$) increases. The result indicates that the mass flow ratio may undermine heat transfer rate of the heat reservoirs (source and sink). Figure 7 shows that the net power output from the ORC is dependent on the mass flow rates of the heat source fluid and the cooling air. Specifically, the net power output increases with increasing mass flow ratio.

Increasing this ratio implies an increase in heat exchanger surface area, therefore, requiring bigger heat exchangers for the ORC plant, it also means cooling fans that require more power will be used at the condenser to supply more air since the system comprises an air cooled condenser. However, the overall net power output increases with increasing the mass flow rate, which equally increases the overall cost of the system; and it connotes that to achieve higher net power output the heat exchanger design is going to prove strenuous as some of the condensing temperatures are quite low. The various working fluids exhibit different characteristics at low temperatures and thus provide more challenges in the design of the ORC system.

4. CONCLUSION

This paper presents a functional analysis of an ORC plant in a hot and humid environment. Twenty one selected working fluids were used in the analysis to find the optimal evaporating temperature (OET) and optimal condensing temperature (OCT). The heat input from the solar-thermal collector is used as the objective functions, which was derived based on thermodynamic theory and obtained from numerical simulations using EES. At mass ratio of 5.0, the R717 (Ammonia) provided the maximum net power output of the ORC at 12.54 (kW) with OET of 364.3 (K) and OCT of 312.4 (K). R717 also had the highest system efficiency at 26.9%. However, the theoretical net power is 12.54 kW; it shows that the utilisation of low grade energy sources (e.g. solar thermal, ocean thermal and biomass waste) have potential for electricity generation. Methanol also provided similar results close to R717 in terms of efficiency and net power output. Methanol is colourless, fairly volatile, has a pungent odour and is highly flammable. R717 however has been widely used in industrial applications and has zero Ozone Depletion Potential (ODP), zero Global Warming Potential (GWP) and minimal environmental impact. R717 is easily detected in the event of leakage due to its sour smell. The results from the thermodynamic analysis of the ORC plant in a hot and humid environment suggest that the wet fluids have better system's figure of merit. Continuous research in this field would favour the development of a sustainable energy system through utilisation of solar driven ORC plant in the sub-Sahara Africa region.

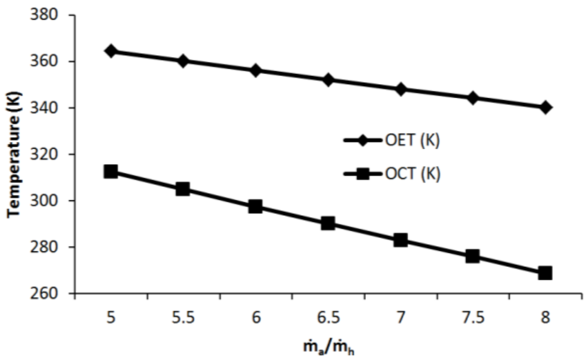

Figure 6: Variation in OCT and OET for R717 with Mass Flow Rate Ratio

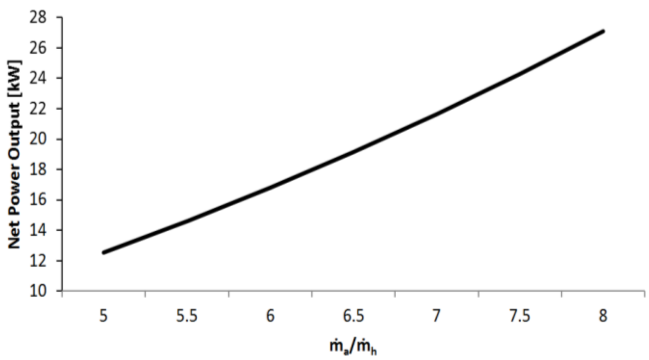

Figure 7: Net Power Output of R717 with Increasing Flow Rate Ratio

5. ACKNOWLEDGEMENT

The publication fee for this work was supported under the CIRCLE Visiting Fellowship programme funded by the UK Department for International Development. Neither the findings nor the views expressed, however, necessarily reflect the policies of the UK Government.

6. REFERENCES

[1] U. Drescher and D. Bru, "Fluid selection for the Organic Rankine Cycle (ORC) in biomass power and heat plants," *Applied Thermal Engineering*, vol. 27, pp. 223–228, 2007.

[2] T. C. Hung, T. Y. Shai, and S. K. Wang, "A review of organic Rankine cycles (ORCs) for recovery of low grade heat," *Energy*, vol. 22, pp. 661–667, 1997.

[3] Y. Goswami, "Energy: the Burning Issue. Renewable Energy Focus, 8 (1), 22-25," *Renew. Energy Focus*, vol. 8, no. 1, pp. 22–25, 2007.

[4] H. Gujba, Y. Mulugetta, and A. Azapagic, "Environmental and economic appraisal of power generation capacity expansion plan in Nigeria," *Energy Policy*, vol. 38, no. 10, pp. 5636–5652, 2010.

[5] V. Khare, S. Nema, and P. Baredar, "Solar-wind hybrid renewable energy system: A review," *Renew. Sustain. Energy Rev.,* vol. 58, pp. 23–33, 2016.

[6] D. K. Lal, B. B. Dash, and a K. Akella, "Optimization of PV / Wind / Micro-Hydro / Diesel Hybrid Power System in HOMER for the Study Area," *Int. J. Electr. Eng. Informatics*, vol. 3, no. 3, pp. 307–325, 2011.

[7] T. Lambert, P. Gilman, and P. Lilienthal, "Micropower System Modeling with Homer," *Integr. Altern. Sources Energy,* pp. 379–418, 2006.

[8] F. Leonforte and C. Del Pero, "Modeling and Performance Monitoring of a Photovoltaic – Thermal (PVT) Water Collector," *Sol. Energy*, vol. 112, pp. 85–99, 2015.

[9] Federal Ministry of Power, "National Renewable Energy and Energy Efficiency Policy," 2015.

[10] R. Siddaiah and R. P. Saini, "A review on planning, configurations, modeling and optimization techniques of hybrid renewable energy systems for off grid applications," *Renew. Sustain. Energy Rev.,* vol. 58, pp. 376–396, 2016.

[11] T. M. de Christo, J. F. Fardin, D. S. L. Simonetti, L. F. Encarnação, and C. E. de Alvarez, "Design and analysis of hybrid energy systems: The Brazilian Antarctic Station case," *Renew. Energy*, vol. 88, pp. 236–246, 2016.

[12] J. E. Boretz, "Supercritical organic Rankine engines," in IECEC '86; *Proceedings of the Twenty-first Intersociety Energy Conversion Engineering Conference*, August 25-29, 1986, pp. 2050–2054.

[13] E. P. Akhator, A. I. Obanor, and D. I. Igbinomwanhia, "Thermal analysis of a small scale solid waste-fired steam boiler for power generation in Benin city , nigeria," *Nigerian Journal of Technology*, Vol. 35, No. 3, pp. 555–561, 2016.

[14] S. Quoilin, M. Van Den Broek, S. Declaye, P. Dewallef, and V. Lemort, "Techno-economic survey of Organic Rankine Cycle (ORC) systems," *Renew. Sustain. Energy Rev.*, vol. 22, pp. 168–186, 2013.

[15] N. Rohmah, G. Pikra, and A. Salim, "Organic Rankine cycle system preliminary design with corn cob biomass waste burning as heat source," *in International Conference on Sustainable Energy Engineering and Application*, 2013, vol. 32, pp. 200–208.

[16] H. Muller-Steinhagen and F. Trieb, "Concentrating solar power – a review of the technology," *Q. R. Acad. Eng.*, vol. Vol.18, pp. 43–50, 2004.

[17] G. Zhaolin and S. Haruki, "Performance of Supercritical cycles for geothermal binary design," *J. Energy Convers. Manag.,* vol. 43, no. 1, pp. 961 – 971, 2002.

[18] B. Saleh, G. Koglbauer, M. Wendl and, and J. Fischer, "Working fluids for low temperature organic Rankine cycles," *Energy,* vol. 32, pp. 1210–1221, 2007.

[19] R. J. Xu and Y. L. He, "A vapour injector-based novel regenerative organic Rankine cycle," *Appl. Energy*, vol. 31, pp. 1238–1243, 2011.

[20] Y. P. Dai, J. F. Wang, and G. Lin, "Parametric optimisation and comparative study of organic Rankine cycle (ORC) for low grade waste heat recovery," *Energy Convers. Manag.,* vol. 50, pp. 576–582, 2009.

[21] C. He, C. Liu, H. Gao, H. Xie, Y. Li, S. Wu, and J. Xu, "The optimal evaporation temperature and working fluids for subcritical organic Rankine cycle," *Energy,* vol. 38, pp. 136–143, 2012.

[22] R. Kobayashi and G. F. Carruth, "Extension to low reduced temperatures of three-parameter corresponding states: vapour pressures, enthalpies and entropies of vaporization, and liquid fugacity coefficients," *Ind. Eng. Chem. Fundam.*, vol. 11, no. 4, pp. 509 – 516, 1972.

[23] Nigerian Meteorological Agency (NIMET), "Port Harcourt meteorological data for 2004 – 2014," 2015.

[24] J. L. Yan, "Thermodynamic principles and formulas for choosing working fluid and parameters in designing power plant of low temperature heat," *J. Eng. Thermodyn.*, vol. 3, no. 1, pp. 1–7, 1982.

[25] Gas Processors Supplier Association, "Air cooled heat exchangers," *in Production Handbook*, Vol. 10, p. 14. 2005.